

Analysis of an electricity–cooling cogeneration system based on RC–ARS combined cycle aboard ship



Youcai Liang, Gequn Shu, Hua Tian, Xingyu Liang, Haiqiao Wei*, Lina Liu

State Key Laboratory of Engines, Tianjin University, 92 Weijin Road, Tianjin 300072, China

ARTICLE INFO

Article history:

Received 12 May 2013

Accepted 28 August 2013

Keywords:

Electricity–cooling cogeneration
Rankine–absorption refrigeration combined cycle
Ship
Fuel consumption

ABSTRACT

In this paper, an electricity–cooling cogeneration system based on Rankine–absorption refrigeration combined cycle is proposed to recover the waste heat of the engine coolant and exhaust gas to generate electricity and cooling onboard ships. Water is selected as the working fluid of the Rankine cycle (RC), and a binary solution of ammonia–water is used as the working fluid of the absorption refrigeration cycle. The working fluid of RC is preheated by the engine coolant and then evaporated and superheated by the exhaust gas. The absorption cycle is powered by the heat of steam at the turbine outlet. Electricity output, cooling capacity, total exergy output, primary energy ratio (PER) and exergy efficiency are chosen as the objective functions. Results show that the amount of additional cooling output is up to 18 MW. Exergy output reaches the maximum 4.65 MW at the vaporization pressure of 300 kPa. The study reveals that the electricity–cooling cogeneration system has improved the exergy efficiency significantly: 5–12% increase compared with the basic Rankine cycle only. Primary energy ratio (PER) decreases as the vaporization pressure increases, varying from 0.47 to 0.40.

© 2013 Elsevier Ltd. All rights reserved.

1. Introduction

As the severity of the energy scarcity and environmental pollution, more attention is paid to search for efficient methods of fuel consumption reduction and emission control on marine diesel engines, which are the main consumers of the fossil fuel and contributors to environment pollution [1,2]. One of the valuable alternative approaches is to capture and reuse the waste heat energy, which is called waste heat recovery (WHR) technology [3].

Before research on waste heat recovery of diesel engine, analysis of energy balance should be first carried out to find out the potential of WHR. A study [4] on a typical two-stroke diesel engine of MAN B&W Diesel had discovered that 25.5% of energy is wasted through the exhaust at ISO ambient reference conditions at 100% Contract Maximum Continuous Rating (CMCR), and 16.5% and 5.6% through the air cooler and jacket water respectively. It is promising to convert waste heat into useful output, which will improve fuel consumption.

In the early time, WHR on board ship was mainly focused on space heating [5], heavy fuel oil heating [6] and ballast water heating [7] and other thermal needs, which were inefficient. With increasingly world-wide concern on energy shorting, many recovery techniques in international combustion engine (ICE) such as turbine, refrigeration, thermoelectric, Rankine cycle and desalina-

tion were carried out [3]. Recently more attentions have been paid for Rankine processes and the number of investigation for optimization of Rankine cycle has increased significantly. Wang et al. [8] presented a review about researches on thermal exhaust heat recovery with RC, including different system configurations, components on performance and working fluids. Teng [9–11] proposed a supercritical Organic Rankine cycle (ORC) system to recover waste heat from heavy-duty diesel engines. Charge air cooler and exhaust gas recirculation (EGR) cooler served as pre-heaters for the working fluid, and the exhaust gas cooler served as an evaporator. Results showed that this system could recover 20% of the waste heat from the engine, making the efficiency of the hybrid energy system over 50%. Srinivasan et al. [12] introduced ORC turbo-compounding in parallel with hot EGR and found 7% improvement in fuel conversion efficiency and 18% reduction in NO_x and CO₂ emission on average. Performances of nine different pure organic working fluids were studied by Wang et al. [13] to find out the optimal working fluid for engine waste heat-recovery. Investigations on ORC system to recover waste heat of diesel engine were also conducted in our previous work, including analytic network process technique used in assessment of WHR methods [14], performance of different configuration [15], selection of working fluids [16,17] and optimal operating parameters [18]. Results showed that ORC WHR system was potential to reduce fuel consumption and improve thermal efficiency. Compared with automobile engine, marine engine onboard ship has some advantages: more stable operation, larger spacing for installing, and larger

* Corresponding author. Tel./fax: +86 022 27891285.

E-mail address: whq@tju.edu.cn (H. Wei).

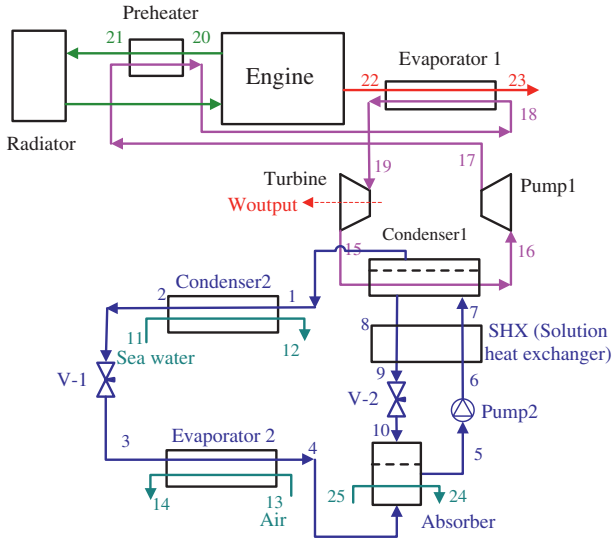


Fig. 1. Schematic of the novel electricity-cool cogeneration system.

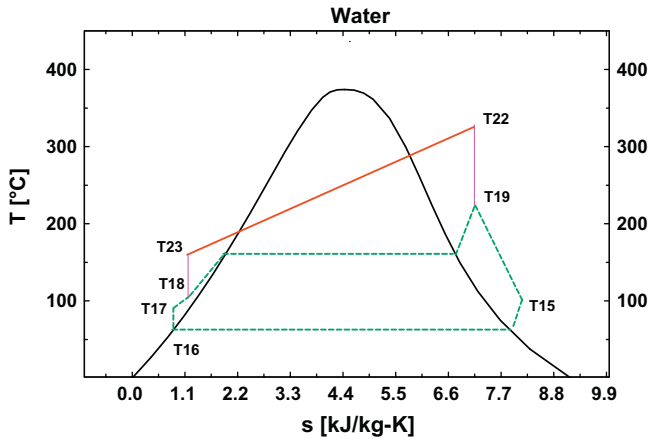


Fig. 2. T-S diagram of working fluid in Rankine cycle.

through a turbine, decreasing the temperature and pressure and generating power. The power is then transferred to the electric generator or output shaft. At last, the steam from the turbine outlet is condensed into saturated fluid again in the condenser, which also acts as an evaporator in the absorption refrigeration cycle. Then a new cycle begins.

In the absorption refrigeration cycle, the heat of the steam at the turbine outlet in RC is used to distill the ammonia vapor from the strong ammonia–water solution in the generator. The rectified ammonia vapor is then condensed through a heat exchanger to replenish the supply of liquid ammonia in the evaporator. Expansion valve V-1 is used to lower the condensation pressure down to the evaporation pressure and control the mass flow rate of the refrigerant. The liquid ammonia is evaporated in a low partial pressure environment, extracting heat from its surroundings and generating cooling power. The vapor out of the evaporator is absorbed in the absorber by weak solution that is depressurized to the evaporation pressure by valve V-2. During this process, the absorber is cooled by the sea water, lowering both the temperature and pressure of absorber, which will make the gaseous ammonia easily dissolve in the weak solution. The strong solution from the absorber is pumped into the solution heat exchanger (SHX) by pump-1 and then into the generator. Then the cycle restarts.

3. Modeling and assumption

The needed thermodynamic parameters, the enthalpies and entropies in different states were obtained by Equation Evaluation Solution (EES), which is usually used in evaluation and optimization of the thermodynamic cycle performance. Computer simulations were also conducted by EES to evaluate the thermodynamic performance of the electricity-cooling combined system. The thermodynamic description of absorption cycle was based on the Schulz's equation of state for the ammonia–water [37] and the relative mathematical model was developed to describe the working process. Before detailed analysis, typical assumptions should be first considered:

- (1) The system is operating in a steady state.
- (2) The efficiency of fluid pumps in the system is 0.8, and that of turbine is 0.7.
- (3) The temperature of the exhaust out of the boiler is higher than the dew point.
- (4) The solution and refrigerant valves are adiabatic.
- (5) Before flowing into the condenser, the water content in the ammonia vapor is assumed to be removed thoroughly in a purification process.
- (6) The refrigerant out of the condenser is saturated liquid and the refrigerant out of the evaporator is saturated vapor.
- (7) The weak solution out of the generator is saturated solution in the corresponding pressure and temperature.
- (8) The simulation program neglects pressure drops and heat losses.

Main operation parameters should also be set before the thermodynamic modeling, as shown in Table 1.

Based on the assumptions and the operation parameters, the working process in each part can be described as follows:

Pump1 (state: 16 → 17):

The fluid pump is driven by electricity and the work consumed by Pump1 is:

$$W_{p1} = \dot{m}(h_{17} - h_{16})/\eta_{p1}$$

Pre-heater (state: 17 → 18):

The pre-heater transfers the heat from the jacket water to the working fluid, enhancing the temperature of the working fluid and relieving energy burden of the evaporator

$$\dot{m}(h_{18} - h_{17}) = c_{pw}\dot{m}_j(T_{20} - T_{21})$$

Evaporator 1 (state: 18 → 19):

This is an isobaric heating process in the evaporator. The heat transferred from the exhaust gas to the working fluid is:

$$\dot{m}(h_{19} - h_{18}) = c_{pg}\dot{m}_g(T_{22} - T_{23})$$

$$Q_{eva1} = c_{pg}\dot{m}_g(T_{22} - T_{23})$$

Turbine (state: 19 → 15):

Non-isentropic expansion process occurs in the turbine. A generator is connected with the turbine, which converts the power into electricity.

$$W = \dot{m}(h_{19} - h_{15})\eta_T\eta_G$$

$$W_{net} = W - W_p$$

$$\dot{m}_{19} = \dot{m}_{15}$$

Condenser1 (state: 15 → 16):

Table 1

Main parameters of operating conditions.

Parameters	Value	Parameters	Value
Atmosphere pressure	101 kPa	Mole fraction of strong solution out of absorber	0.8
Atmosphere temperature	303 K	Min temperature difference in the condenser of ARS	6 K
Sea water temperature	288 K	Min temperature difference in the evaporator of ARS	5 K
Efficiency of pump	0.8	Effectiveness of SHX	0.9
Efficiency of electric generator	0.8	Min temperature difference in the absorber	6 K
Efficiency of turbine	0.7	Condensation temperature of refrigeration	294 K
Exhaust gas temperature of evaporator outlet	373 K	Set temperature for the refrigerated environment	273 K
Jacket coolant temperature of pre-heater inlet	363 K	Min temperature difference in the pre-heater	5 K

This is an isobaric condensation process in the condenser. The energy transferred in this process can be expressed as:

$$Q_{cond1} = \dot{m}(h_{15} - h_{16})$$

There are eight main components in the absorption refrigeration cycle: generator, condenser, two expansion valves, evaporator, absorber, fluid pump and SHX. The working process in each part can be described as follows:

Generator (7 → 1, 8):

The ammonia–water absorbs heat energy in the generator. The energy balance and quality balance can be expressed as:

$$\dot{m}(h_{15} - h_{16}) = \dot{m}_1 h_1 + \dot{m}_8 h_8 - \dot{m}_7 h_7$$

$$\dot{m}_7 = \dot{m}_8 + \dot{m}_1$$

$$\dot{m}_7 a = \dot{m}_8 b + \dot{m}_1 c$$

Condenser2 (1 → 2):

In the condenser 2, the seawater is used as the heat sink. The energy balance and quality balance can be expressed as:

$$\dot{m}_1 h_1 - \dot{m}_2 h_2 = \dot{m}_{12} h_{12} - \dot{m}_{11} h_{11}$$

$$\dot{m}_1 = \dot{m}_2$$

Expansion valve (2 → 3)

The expansion valve is used to control the evaporation pressure as well as the fluid mass flow rate. It can be seen as an adiabatic process.

$$h_2 = h_3$$

$$\dot{m}_3 = \dot{m}_2$$

Evaporator 2 (3 → 4):

The evaporator absorbs heat from environment and generates refrigeration.

$$\dot{m}_4 h_4 - \dot{m}_3 h_3 = c_{pa} \dot{m}_a (T_{13} - T_{14})$$

$$\dot{m}_4 = \dot{m}_3$$

$$Q_{eva2} = c_{pa} \dot{m}_a (T_{13} - T_{14})$$

Absorber (4, 10 → 5):

The ammonia vapor flows into the absorber and is absorbed by the weak solution.

$$\dot{m}_4 h_4 + \dot{m}_{25} h_{25} + \dot{m}_{10} h_{10} = \dot{m}_5 h_5 + \dot{m}_{24} h_{24}$$

$$\dot{m}_4 + \dot{m}_{10} = \dot{m}_5$$

$$\dot{m}_4 c + \dot{m}_{10} b = \dot{m}_5 a$$

$$\dot{m}_{24} = \dot{m}_{25}$$

Pump2 (5 → 6):

The fluid pump is driven by electricity. The electricity consumed by Pump-1 can be expressed as:

$$W_{p2} = (\dot{m}_6 h_6 - \dot{m}_5 h_5) / \eta_{p2}$$

$$\dot{m}_6 = \dot{m}_5$$

SHX (6 → 7, 8 → 9):

SHX is also named economizer, which can improve the thermal efficiency of cooling cycle [38]. The energy balance and quality balance can be expressed as [36,38]:

$$\dot{m}_7 h_7 - \dot{m}_6 h_6 = \dot{m}_8 h_8 - \dot{m}_9 h_9$$

$$\dot{m}_7 = \dot{m}_6$$

$$\dot{m}_9 = \dot{m}_8$$

$$T_9 = \varepsilon T_6 + (1 - \varepsilon) T_8$$

Solution valve V-2 (9 → 10):

As the solution valve is adiabatic, the following equation can be expressed:

$$h_9 = h_{10}$$

$$\dot{m}_9 = \dot{m}_{10}$$

Coefficient of performance (COP) is adopted as the performance function of refrigeration cycle. COP is defined as cooling load at the evaporator (Q_{eva2}) divided by energy input at the generator (Q_{gen}). In order to calculate the net COP, power consumption by the pump is taken into account.

$$COP = Q_{eva2} / (Q_{cond1} + W_{p2} / \eta_{p2})$$

In this study, primary energy rate (PER) [39] and exergy efficiency are used as criterions to evaluate the combined cycle performance of electricity and cooling. The PER is defined as the ratio of the energy output to the energy input the combined system as follows:

$$PER = \frac{Q_{output}}{Q_{input}} = \frac{W_{out} + Q_{eva2}}{Q_{pre} + Q_{eva1} + W_{p1} + W_{p2}}$$

There are huge differences between the energy grade of electricity and cooling effect. While considering the difference of different energy grade, exergy efficiency is often used to evaluate the performance of cogeneration system. Exergy efficiency of cogeneration system is generally defined as the ratio of exergy output to the exergy input to the cycle [40–42]:

$$\eta_{II} = \frac{E_{output}}{E_{input}} = \frac{W_{net} + E_c}{E_{input}}$$

$$E_c = \left(\frac{T_0}{T} - 1 \right) Q_{eva2}$$

Wherein, the η_{II} is the exergy efficiency, E_{input} is the exergy of heat source, W_{net} is the electricity output of generator, and E_c is

the exergy of cooling capacity, which is defined as the maximum useful work attainable from a heating process between the system (T_0) and the reference environment (T) [42].

4. Model validations

As there is no relevant suitable model to validate such configuration of the whole cogeneration system, the RC validation and absorption refrigeration cycle validation were conducted separately. The mass and energy balance of steam RC was based on the Ref. [44]. The errors of main parameters were shown in Table 2, in which the maximum error is about 5.5%. The thermodynamic description of absorption cycle was based on the Schulz's equation of state for the ammonia–water [37] and the validation of mathematical model was based on Ref. [36]. The thermodynamic properties at various states in absorption refrigeration cycle were shown in Table 3. We can learn that the maximum error is less than 9.38%. For the performance of the absorption refrigeration cycle, the COP is 0.6552 compared with 0.6160 in Ref. [36], and the error is about 6.36%. Such a deviation is with sufficient accuracy for our investigation.

5. Results and discussion

As the condenser of RC is also the evaporator of the absorption refrigeration cycle, the temperature and mass flow rate of the working fluid in it are important factors that influence the whole system. In this investigation, superheat of the working fluid and vaporization pressure of Rankine cycle are set as variables. Condensation temperature of RC should not be too high to avoid the case that the temperature of working fluid at the outlet of Pump 1 is higher than that of the jacket water in the pre-heater. The simulation shows that the condensation temperature of Rankine cycle should be lower than 358 K. Moreover, the superheat should be higher to avoid the liquid-hit problem of the turbine blade. The superheat of steam is found to range from 100 K to 180 K. In addition, the cooled exhaust temperature is set to be above 100 °C [43], to avoid the low-temperature corrosion of the exhaust pipe due to the acid dew point.

As water is a wet working fluid, it is important to make sure that the steam at the turbine outlet is dry to prevent the turbine from liquid-hit. Control of the mass flow rate of the working fluid and the vaporization pressure of RC can avoid the above issue for a fixed heat source temperature (314 °C). In the presented simula-

tion work, the mass flow rate working fluid depends on the temperature of the heat source by means of the pinch point method [44]. In this paper, the quality of the steam at steam turbine outlet is controlled by the vaporization pressure of RC. For a given superheat and condensation temperature, a maximum vaporization pressure exists. If the working pressure is higher than this maximum vaporization pressure, the steam at the turbine outlet will partially become liquid, which does harm to turbine blade.

In Fig. 3, it is evident that the maximum cycle pressure of RC increases with the condensation temperature and superheat of the working fluid. As the superheat and condensation temperature of working fluid increase, the maximum vaporization pressure of RC increases. For wet working fluid water, as the superheat of steam increases, the state point at the turbine outlet goes far away from the saturated vapor curve, resulting in the maximum vaporization pressure available also increasing. Meanwhile, with the increase of the condensation temperature, the state of saturated vapor curve of the condensation process moves towards left continuously, which allows a higher pressure for RC.

Taking the limitation of the dew point into consideration, the exhaust temperature cannot be too low for the prevention of exhaust pipe corrosion. Therefore, the following simulation results are based on the condensation temperature of 323 K and the pressure ranges with the maximum vaporization pressure provided in Fig. 3.

The electricity output is a significant important parameter to evaluate the performance of RC. In Fig. 4 the electricity output is plotted with respect to the vaporization pressure of RC. It is evident that when the superheat is lower than 160 °C, the electricity power increases with the increase of RC vaporization pressure; when the superheat is higher than 160 °C, the electricity increases at first and then decreases as the RC vaporization pressure increases. The maximum electricity output exists at about 700 kPa. This is

Table 2
Error analysis of performance of Rankine cycle.

Rankine cycle	Presented study	Ref. [44]	Error (%)
T_{19} (K)	573.7	573.2	0.08
$\eta_{th,ORC}$	0.1952	0.185	5.5
$P_{turbine}$ (kW)	128.2	135.3	5.2
P_{net} (kW)	127.8	132.7	3.7
P_{cond} (kPa)	69.72	70.14	0.6

Table 3
Error analysis of thermodynamic properties at various states in absorption refrigeration cycle.

Fluid state	T (°C)	T (°C) in Ref. [36]	P (kPa)	P (kPa) in Ref. [36]	Concentration (%)	Concentration in Ref. [36] (%)
Generator NH3 exit (1)	100	100	1167	1166.92	100	100
Condenser NH3 exit (2)	30	30	1167	1166.92	100	100
Evaporator NH3 exit (4)	−5	−5	352.8	354.42	100	100
Absorber solution exit (5)	25	25	352.8	354.42	53.5	52.24
Generator solution inlet (7)	64.2	67.7	1167	1166.92	53.5	52.24
Generator solution exit (8)	100	100	1167	1166.92	36.7	33.55
Absorber solution inlet (10)	40	40	352.8	354.42	36.7	33.55

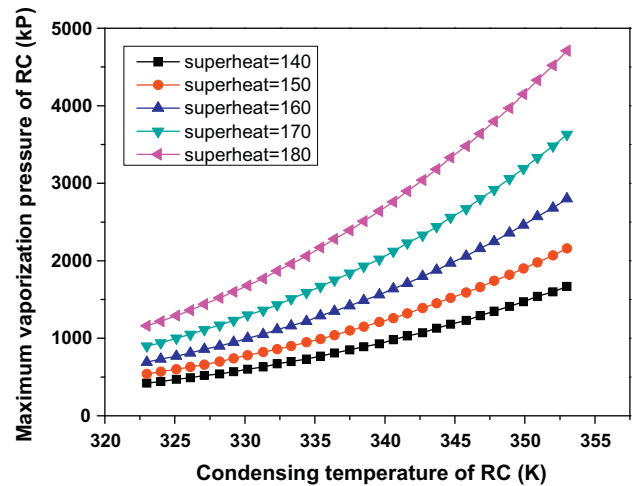


Fig. 3. Influence of condensation temperature on the saturated steam pressure.

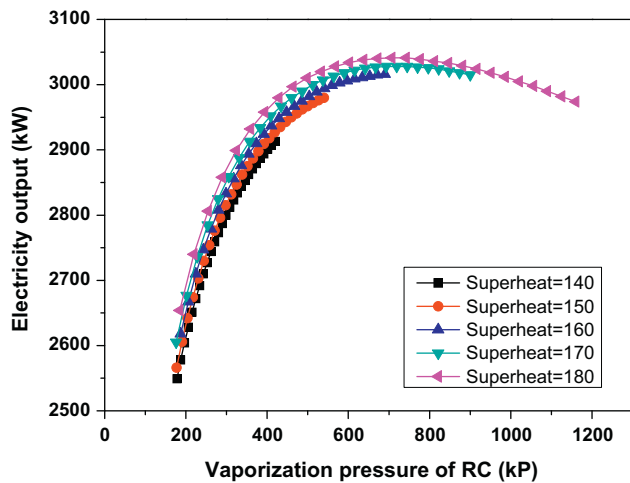


Fig. 4. Influence of vaporization pressure of RC on the electricity output.

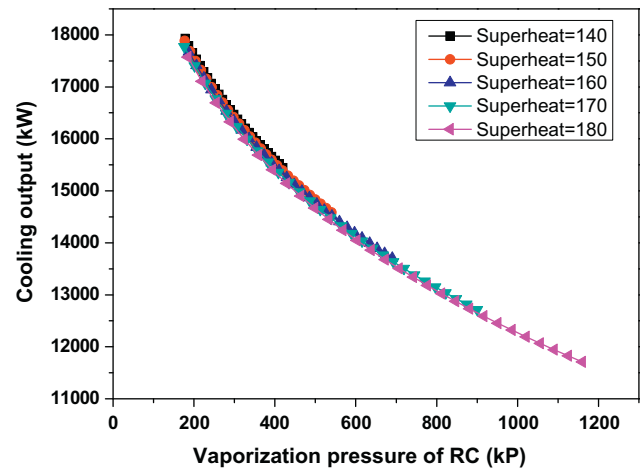


Fig. 5. Influence of vaporization pressure of RC on cooling output.

mainly because that the electricity output depends on both the working fluid mass flow rate and the enthalpy difference between turbine inlet and outlet. As the vaporization pressure increases, the enthalpy difference increases. However, the mass flow rate of steam decreases. According to the T-S diagram, when the vaporization pressure increases, the temperature of saturated liquid point, which is also the pinch point between the working fluid and the exhaust, also increases. For a given pinch point temperature difference, the temperature of exhaust gas from the evaporator outlet also increases, meaning that the total heat transferred from the exhaust to the working fluid decreases. In addition, with the increase of vaporization pressure, the working fluid temperature at turbine inlet or evaporator outlet also increases. Though with the increase of vaporization pressure, the working fluid temperature at evaporator inlet also increases, whose effect is slighter than the increase of the temperature at evaporator outlet. The enthalpy increment in the evaporator increases. So the mass flow rate of the steam decreases. These two factors lead to an optimum vaporization pressure existing. Therefore, the vaporization pressure of RC should be selected in range of 600–800 kPa.

Cooling capacity is an important factor that evaluates a cooling system's ability to remove heat. From Fig. 5 it can be observed that the cooling capacity decreases with the increase of the vaporization pressure of RC. In this simulation, the condensation temperature and evaporation temperature of ARS is fixed. Therefore, the cooling capacity of the absorption cycle depends on the mass flow rate of ammonia vapor separated in the generator. In the absorption refrigeration cycle, the generating temperature influences significantly on the ammonia concentration of the weak ammonia–water solution in the outlet of generator. For a higher generating temperature (corresponding to the condensation temperature of RC), the ammonia vapor separated from per unit mass of ammonia–water solution is more than that of low generating temperature. However, the mass flow rate of ammonia–water solution into the generator decreases. In the case of a lower generating temperature, although the ammonia vapor separated from per unit mass of ammonia–water solution is less, the mass flow rate of strong ammonia–water solution increases. Based on the simulation results, as the vaporization pressure of RC increases, the ammonia vapor separated from solution become less, which leads to a lower cooling capacity of refrigeration cycle. Based on the analysis above, a conclusion can be achieved that it is effective to increase the mass flow rate of ammonia–water solution properly to enhance the cooling capacity.

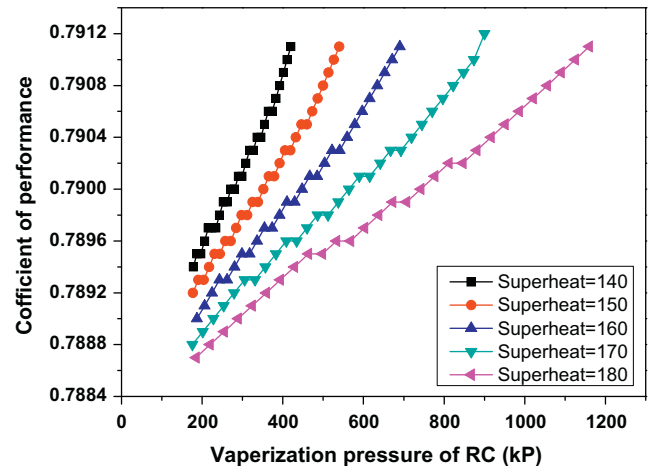


Fig. 6. Influence of vaporization pressure of RC on COP.

The COP is defined as the cooling capacity divided by the energy input in refrigeration cycle. The curves in Fig. 6 refer to the COP of refrigeration cycle at different vaporization pressures. It can be noted that COP increases sharply with the vaporization pressure increasing. The slope of the curve is greater for a lower superheat. As mentioned above, the cooling capacity decreases as the vaporization pressure of RC increases. However, the condensation temperature must be relatively high to avoid the liquid-hit problem of the turbine blade, resulting in the mass flow rate of ammonia–water decreasing significantly. Therefore, the heat energy transferred from RC to ARS becomes smaller. It is the reason why the COP increases with the increasing vaporization pressure of RC, although the cooling capacity reduces.

Primary energy ratio (PER) is simply the ratio of the useful energy output (electricity and cooling power output in this study) to the necessary energy input. This ratio is a measure of the overall efficiency of this electricity–cooling cogeneration system, taking into account the energy losses related to the generation of electricity and cooling. As mentioned above, the variation of RC vaporization pressure significantly influences on electricity output, cooling capacity as well as heat energy used in this combined system. The value of cooling production in the cogeneration system is much more than the electricity generation. For the same superheat, the variation trend of PER is the same as the cooling capacity and decreases with the increasing vaporization pressure, although the

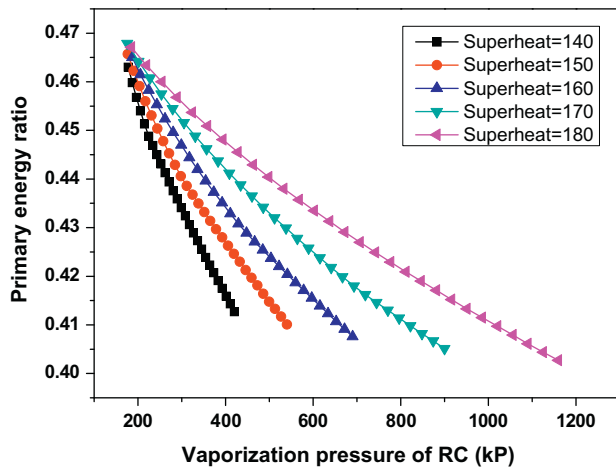


Fig. 7. Influence of vaporization pressure of RC on PER.

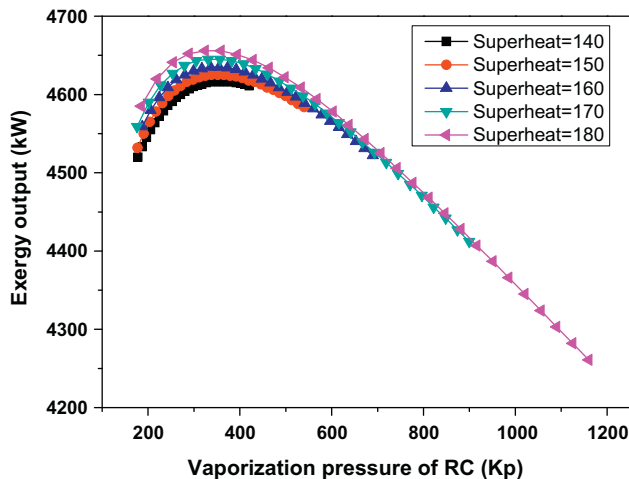


Fig. 8. Influence of vaporization pressure of RC on total exergy output.

electricity increases. We can come to a conclusion from Fig. 5 that the superheat has little effect on the cooling capacity. For different superheat, the mass flow rate of working fluid is less for a higher superheat, in which less heat energy will be transferred from exhaust gas and jacket water to the working fluid of RC. As a result, the PER obtains a higher value for a given vaporization pressure of RC (see Fig. 7).

Variations of exergy output of the cogeneration system at different RC vaporization pressures are shown in Fig. 8. Exergy output is the value of electricity exergy plus cooling exergy. This parameter indicates the maximum useful energy available aboard ships, which takes energy grade into consideration. It can be observed that the exergy output increases at first and then decreases. A maximum value of exergy output of cogeneration system exists at the vaporization pressure of 350 kPa. In addition, a higher superheat can produce a slightly higher exergy. From the results mentioned above, the electricity increases at first and then decreases when the vaporization pressure is over 700 kPa. However, the cooling capacity decreases in all the range. Therefore, the parabolic trend still exists while the maximum value occurs ahead. According to the marine engine power output of 51,480 kW, the useful power of this system is possible to enhance 9%.

Results obtained in Fig. 9 clearly show the benefits of the cogeneration system compared to the basic Rankine cycle only. In this

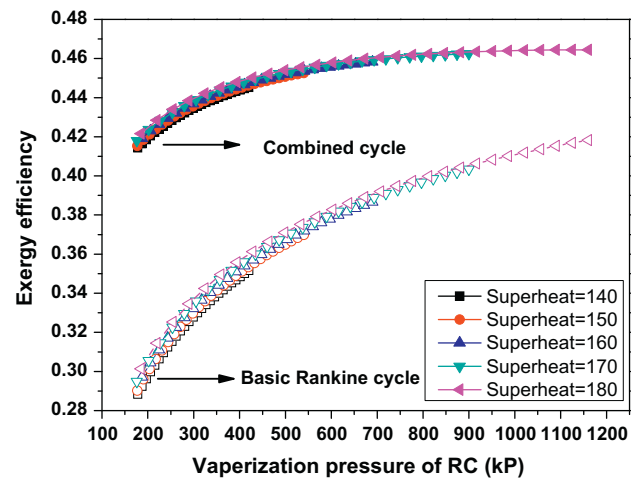


Fig. 9. Comparison of exergy efficiency between basic Rankine cycle and the novel cogeneration cycle.

cogeneration system, the object is to recover the waste heat from the marine engine and convert it into electricity and cooling. The total output of these two parameters determines the performance of the cogeneration system. It can be observed in Fig. 9 that the cogeneration system has a significant improvement of the exergy efficiency, especially at a lower vaporization pressure of RC. At the vaporization pressure of 150 kPa, the exergy efficiency increases from 0.2883 to 0.4143; at the vaporization pressure of 1160 kPa, the exergy efficiency increases from 0.4183 to 0.4644. The cogeneration system improves its exergy output efficiency in the range of 4.61–12.6%. All these results indicate that this cogeneration system is potential to enhance the performance of the WHR system.

6. Conclusions

In this investigation, simulation of a novel electricity–cooling cogeneration system based on RC and absorption refrigeration cycle is presented. The combined cycle is driven by the waste heat of a two-stroke marine engine. The working fluid of RC is water, which is first preheated by the jacket water of engine and then evaporated and superheated by the exhaust gas. Ammonia–water solution is used as the working pair in absorption refrigeration cycle, which is driven by the heat of steam at the turbine outlet in RC. Model calculations were performed at different vaporization pressures of RC. Some conclusions can be achieved as the following:

- (1) The proposed electricity–cooling cogeneration system is possible to enhance the useful output of the whole power plant by about 9%.
- (2) The proposed cogeneration system can satisfy the electricity and cooling demands simultaneously. The maximum electricity output is about 3.05 MW at the vaporization pressure of 700 kPa; the maximum cooling capacity is about 18 MW at the vaporization pressure of 150 kPa.
- (3) Compared with the basic RC, application of the cogeneration system is possible to increase the exergy efficiency of WHR system from 0.2883 to 0.4143, which increases by 43.7% relatively.

According to the results mentioned above, the performance of the WHR system can be improved significantly by adding an absorption refrigeration cycle, which possesses great potential to recover the waste heat energy of marine engine. In addition,

investigation on energy balance and energy management will be carried out in future to further validate its effective application.

Acknowledgements

This work was supported by the National Nature Science Foundation of China (No.51206117), the National Basic Research Program of China (973Program, No.2011CB707201) and Natural Science Foundation of Tianjin (No.12JCQNJC04400). The author gratefully acknowledge them for support of this work.

References

- [1] Capaldo K, Corbett JJ, Kasibhatla P, Fischbeck PS, Pandis SN. Effect of ship emissions on sulphur cycling and radiative climate forcing over the ocean. *Nature* 1999;400:743–6.
- [2] Tzannatos E. Ship emissions and their externalities for Greece. *Atmos Environ* 2010;44:2194–202.
- [3] Shu G, Liang Y, Wei H, Tian H, Zhao J, Liu L. A review of waste heat recovery on two-stroke IC engine aboard ships. *Renew Sustain Energy Rev* 2013;19:385–401.
- [4] MAN B&W Diesel LTD. Thermo Efficiency System (TES) for reduction of fuel consumption and CO₂ emission. Copenhagen, Denmark: MAN B&W Diesel A/S; July 2005.
- [5] Bidini G, Maria F, Generosi M. Micro-cogeneration system for a small passenger vessel operating in a nature reserve. *Appl Therm Eng* 2005;25:851–65.
- [6] Tien WK, Yeh RH, Hong JM. Theoretical analysis of cogeneration system for ships. *Energy Convers Manage* 2007;48:1965–74.
- [7] Rigby GR, Hallegraef GM, Sutton C. Novel ballast water heating technique offers cost-effective treatment to reduce the risk of global transport of harmful marine organisms. *Mar Ecol Prog Ser* 1999;191:289–93.
- [8] Wang T, Zhang Y, Peng Z, Shu G. A review of researches on thermal exhaust heat recovery with Rankine cycle. *Renew Sustain Energy Rev* 2011;15:2862–71.
- [9] Teng H, Regner G, Cowland C. Achieving high engine efficiency for heavy-duty diesel engines by waste heat recovery using supercritical organic-fluid Rankine cycle. SAE paper, No. 2006-01-3522; 2006.
- [10] Teng H, Regner G, Cowland C. Waste heat recovery of heavy-duty diesel engines by organic Rankine cycle Part I: Hybrid energy system of diesel and Rankine engines. SAE paper, No. 2007-01-0537; 2007.
- [11] Teng H, Regner G, Cowland C. Waste heat recovery of heavy-duty diesel engines by organic Rankine cycle Part II: Working fluid for WHR-ORC. SAE paper, No. 2007-01-0543; 2007.
- [12] Srinivasan KK, Mago PJ, Krishnan SR. Analysis of exhaust waste heat recovery from a dual fuel low temperature combustion engine using an Organic Rankine Cycle. *Energy* 2010;35:2387–99.
- [13] Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. *Energy* 2011;36:3406–18.
- [14] Liang X, Sun X, Shu G, Sun K, Wang X, Wang X. Using the analytic network process (ANP) to determine method of waste energy recovery from engine. *Energy Convers Manage* 2013;66:304–11.
- [15] Shu G, Zhao J, Tian H, Liang X, Wei H. Parametric and exergetic analysis of waste heat recovery system based on thermoelectric generator and organic Rankine cycle utilizing R123. *Energy* 2012;45:806–16.
- [16] Tian H, Shu G, Wei H, Liang X, Liu L. Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE). *Energy* 2012;47:125–36.
- [17] Shu G, Liu L, Tian H, Wei H, Xu X. Performance comparison and working fluid analysis of subcritical and transcritical dual-loop organic Rankine cycle (DORC) used in engine waste heat recovery. *Energy Convers Manage* 2013;74:35–43.
- [18] Yu G, Shu G, Tian H, Wei H, Liu L. Simulation and thermodynamic analysis of a bottoming Organic Rankine Cycle (ORC) of diesel engine (DE). *Energy* 2013;51:281–90.
- [19] Wang SG, Wang RZ. Recent developments of refrigeration technology in fishing vessels. *Renew Energy* 2005;30:589–600.
- [20] Srihirin P, Aphornratana S, Chungpaibulpatana S. A review of absorption refrigeration technologies. *Renew Sustain Energy Rev* 2001;5:343–72.
- [21] Fernandez-Seara J, Vales A, Vazquez M. Heat recovery system to power an onboard NH₃–H₂O absorption refrigeration plant in trawler chiller fishing vessels. *Appl Therm Eng* 1998;18:1189–205.
- [22] Manzela AA, Hanriot SM, Cabezas-Gomez L. Using engine exhaust gas as energy source for an absorption refrigeration system. *Appl Energy* 2010;87:1141–8.
- [23] Abusoglu A, Kanoglu M. First and second law analysis of diesel engine powered cogeneration systems. *Energy Convers Manage* 2008;49:2026–31.
- [24] Tamura I, Taniguchi H, Sasaki H, Yoshida R, Sekiguchi I, Yokogawa M. An analytical investigation of high-temperature heat pump system with screw compressor and screw expander for power recovery. *Energy Convers Manage* 1997;38:1007–13.
- [25] Goswami DY. Solar thermal technology: present status and ideas for the future. *Energy Sources* 1998;20:137–45.
- [26] Goswami DY, Xu F. Analysis of a new thermodynamic cycle for combined power and cooling using low and mid temperature solar collectors. *ASME J Sol Energy Eng* 1999;121:91–7.
- [27] Hasan AA, Goswami DY. Exergy analysis of a combined power and refrigeration thermodynamic cycle driven by a solar heat source. *ASME J Sol Energy Eng* 2003;125:55–60.
- [28] Lu S, Goswami DY. Optimization of a novel combined power/refrigeration thermodynamic cycle. *ASME J Sol Energy Eng* 2003;125:212–7.
- [29] Tamm G, Goswami DY, Lu S, Hasan AA. Novel combined power and cooling thermodynamic cycle for low temperature heat sources, Part 1: Theoretical investigation. *ASME J Sol Energy Eng* 2003;125:218–22.
- [30] Tamm G, Goswami DY. Novel combined power and cooling thermodynamic cycle for low temperature heat sources, Part 2: Experimental investigation. *ASME J Appl Mech* 2003;125:223–9.
- [31] Vijayaraghavan S, Goswami DY. On evaluating efficiency of a combined power and cooling cycle. *ASME J Energy Resour Technol* 2003;125:221–7.
- [32] Zheng D, Chen B, Qi Y, Jin H. Thermodynamic analysis of a novel absorption power/cooling combined-cycle. *Appl Energy* 2006;83:311–23.
- [33] Sulzer RTA 96C. Engine selection and project manual. Wartsila; June 2001.
- [34] Dzida M. On the possible increasing of efficiency of ship power plant with the system combined of marine diesel engine, gas turbine and steam turbine, at the main engine-steam turbine mode of cooperation. *Polish Maritime Res* 2009;16(1(59)):47–52.
- [35] Vaezi M, Passandideh-Fard M, Moghiman M, Charmchi M. Gasification of heavy fuel oils: a thermochemical equilibrium approach. *Fuel* 2011;90:878–85.
- [36] Sun DW. Comparison of the performances of NH₃–H₂O, NH₃–LiNO₃, and NH₃–NaSCN absorption refrigeration systems. *Energy Convers Manage* 1998;39:357–68.
- [37] Schulz SCG. Equations of state for the system ammonia–water for use with computers. Washington DC; 1971 [Reprint from the proceeding of the XIIth International Congress of Refrigeration, 2].
- [38] Adewusi SA, Zubair SM. Second law based thermodynamic analysis of ammonia–water absorption systems. *Energy Convers Manage* 2004;45:2355–69.
- [39] Kong XQ, Wang RZ, Huang XH. Energy efficiency and economic feasibility of CCHP driven by stirling engine. *Energy Convers Manage* 2004;45:1433–42.
- [40] Ahmadi P, Dincer I, Rosen MA. Exergo-environmental analysis of an integrated organic Rankine cycle for trigeneration. *Energy Convers Manage* 2012;64:447–53.
- [41] Huangfu Y, Wu JY, Wang RZ, Kong XQ, Wei BH. Evaluation and analysis of novel micro-scale combined cooling, heating and power (MCCHP) system. *Energy Convers Manage* 2007;48:1703–9.
- [42] Balli O, Aras H, Hepbasli A. Thermodynamic and thermoeconomic analyses of a trigeneration (TRIGEN) system with a gas–diesel engine: Part I – Methodology. *Energy Convers Manage* 2010;51:2252–9.
- [43] Zhang HG, Wang EH, Fan BY. Heat transfer analysis of a finned-tube evaporator for engine exhaust heat recovery. *Energy Convers Manage* 2001;42:438–47.
- [44] Siddiqi MA, Atakan B. Alkanes as fluids in Rankine cycles in comparison to water, benzene and toluene. *Energy* 2012;45:256–63.